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RUNE BRINCKER & J. D. SØRENSEN

HIGH-SPEED STOCHASTIC FATIGUE TESTING

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High-speed Stochastic Fatigue Testing

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ABSTRACT

Good stochastic fatigue tests are difficult to perform. One of the major reasons is that ordinary servo hydraulic loading systems realize the prescribed load history accurately at very low testing speeds only. If the speeds used for constant amplitude testing are applied to stochastic fatigue testing, quite unacceptable errors are introduced. Usually this problem is solved by running the tests at very low speeds and by editing the load history in order to reduce the duration of the test. In this paper a new method for control of stochastic fatigue tests is proposed. It is based on letting the analog control device remain as the basic control mechanism in the system, but distorting the input signal by computer in order to minimize the errors of the load history extremes. The principle proves to be very efficient to reduce all kinds of system errors and has shown to be able to increase the allowable speed by a factor from 10 to 30.

1 INTRODUCTION

In structural engineering most of the fatigue problems are of a distinct stochastic nature. Typical examples are wind loaded structures like chimneys, towers and slender bridges, or wave loaded structures e.g. offshore platforms.

Even though fracture mechanics have added to the understanding of the fatigue problem, the basic mechanisms of fatigue are not fully understood, and up till now no single theory has been formulated to give a full description of the fatigue damage accumulation. The frequently used relationship between crack growth rate and stress intensity range given by Paris' law, have shown, even in the constant amplitude case, to give an improper description of test results, Ditlevsen & Olesen [2]. However, the stochastic case is much more complicated to deal with. Some attempts have been made to establish a theory for history dependency using a crack closure model together with Paris' law or other alikes, Schijve [1]. This has proved successful in understanding some of the effects of acceleration and deceleration of crack growth, but the conclusion still are, that reliable theories of stochastic fatigue are missing, even at the Paris' law level for the constant amplitude problem.

Moreover, the stochastic fatigue problem has not been studied very much experimentally. There is a great need for systematic experimental investigations in the area, but there are several problems that must be overcome before good experiments can be carried out. Two problems are of major concern: the problem of simulation of load histories, and the problem of testing speed and loading errors. The first problem is treated in Sørensen & Brincker [3], and the second problem is the subject of this paper.

2 ERRORS IN LOADING SYSTEMS

Nearly all modern equipment used for fatigue testing are hydraulic servosystems as shown i figure 1.

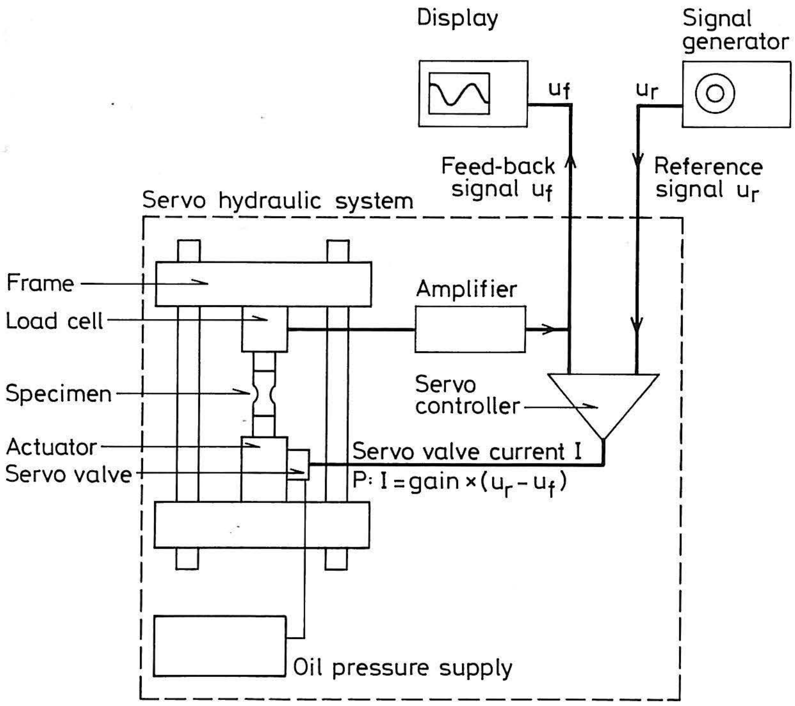

Figure 1. Typical servo hydraulic setup.

The system is fed by a command signal or reference signal u_r which is normally a picture of the load history to be experienced by the specimen. The actual load on the specimen is reflected by the feedback signal u_f. Because of the principles of classical servo control, however, the specimen will never experience the loadings indicated by the reference signal. The oil flow to the actuator is determined by errors in the system, i.e. deviations in feedback from the desired value given by the reference signal. For instance, in proportional control mode (P) the oil flow is proportional to the error. Usually also differential (D) and integral (I) terms will be used to establish a full PID control and thereby improve system response, but the errors will never totally vanish.

A simple way to quantify the system errors for a given setup and a given specimen, is to measure the frequency response function, FRF, for the system. The system is loaded with a harmonic reference signal

$$u_r(t) = u_r^0 e^{i2\pi\nu t} \tag{1}$$

where t is the time, ν is the cyclic frequency, and u_r^0 is the amplitude. Then the response

$$u_f(t) = u_f^0 e^{i(2\pi\nu t+\varphi)} \tag{2}$$

can be measured and the FRF $H(\nu)$ given by

$$H(\nu) = \frac{u_f^0}{u_r^0} e^{i\varphi} \tag{3}$$

is obtained. Both the size of the FRF $|H(\nu)|$, or better $1 - |H(\nu)|$, and the phase $\varphi(\nu)$ are measures of the system errors.

A typical FRF for a well tuned and well-maintained servo hydraulic setup is shown in figure 2. The system shows a behaviour like a strongly damped dynamic system with one degree of freedom.

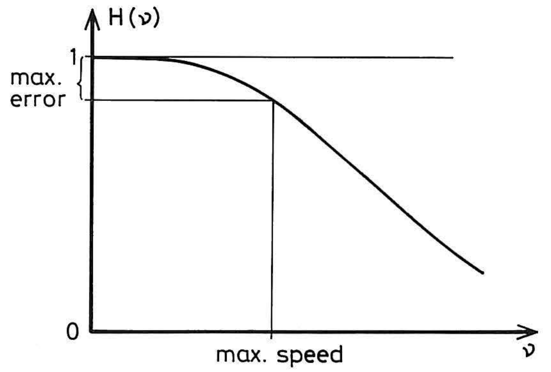

Figure 2. Qualitative behaviour of servo hydraulic loading system.

In the case of constant amplitude testing it is easy to correct for system errors. One simply adjusts the offset and the amplitude of the reference signal to achieve the desired feedback signal. In stochastic fatigue a similar procedure is not available, so once the acceptable errors are established, the maximum speed must be obtained directly from the FRF as shown in figure 2.

It is in fact a very difficult problem to correct for the system errors in the case of fully stochastic loadings. It is difficult because the system errors are not described by one FRF. In fact the FRF for the system depends on the specimen, the damage of the specimen, the level of maintenance of the system, the oil temperature, the tuning of the system (tuning of servo valve and servo amplifier) and the load level. This means that because of drift, changes of the specimen characteristics and non-linearities in the system, one has to choose a new FRF for every load cycle in the loading process. Of course this is not expedient.

Some of the factors mentioned above have a rather strong influence on the system behaviour. The influence of non-linear effects, of level of maintenance (oil leak) and of system tuning upon system behaviour are illustrated in figure 3, 4 and 5.

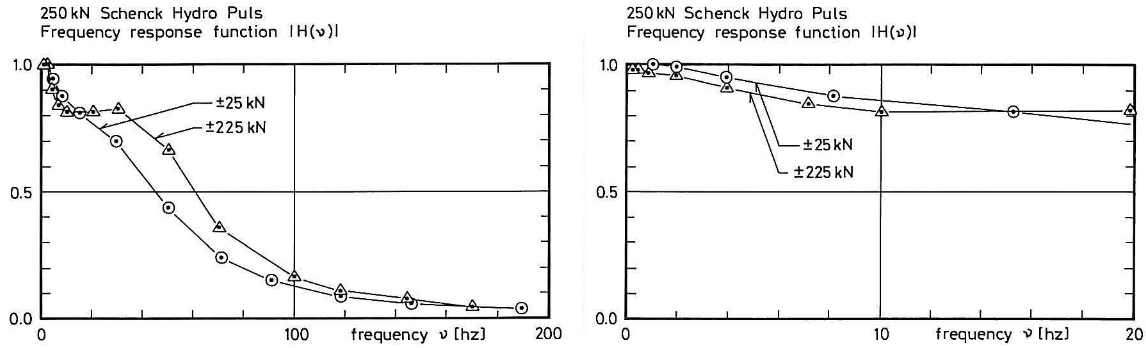

Figure 3. Nonlinearities of a typical system.

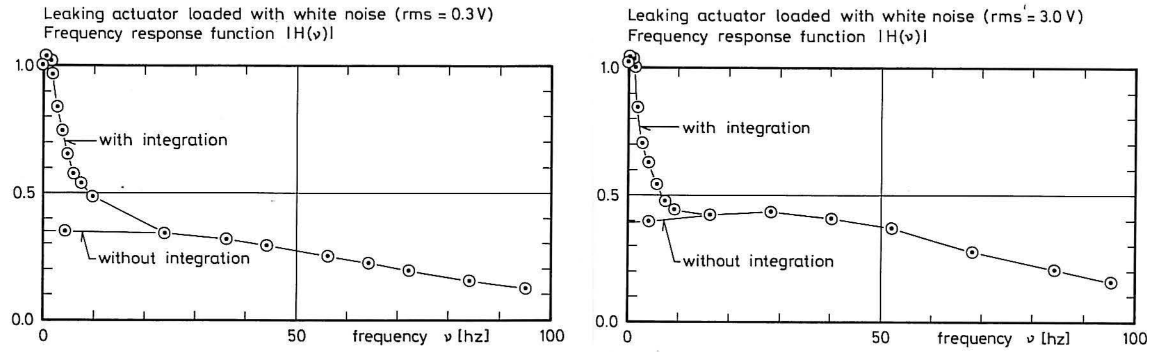

Figure 4. Influence of maintenance level.

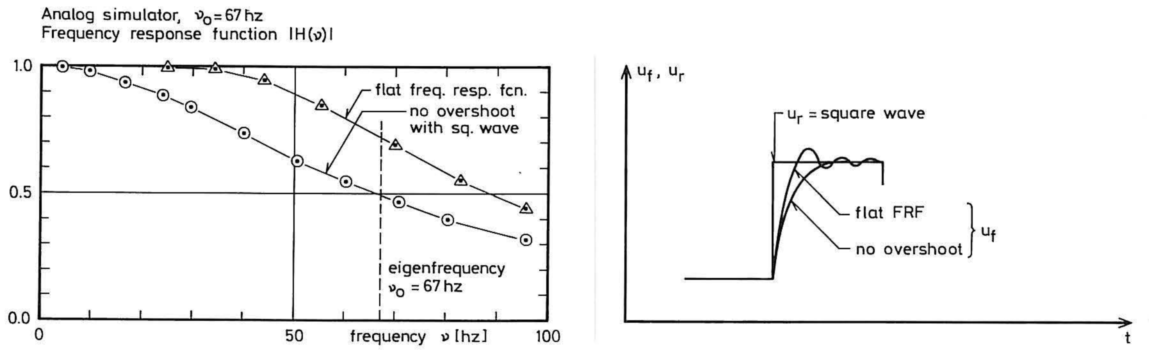

Figure 5. Influence of tuning procedure.

The FRFs shown on figure 3 were measured using a harmonic input to a new Schenck Hydropuls testing machine with a 250 kN actuator. The actuator and the load cell were joined directly, corresponding to the case of an infinitely rigid specimen, and optimal system response was assured by following the recommended automatic tuning procedure just prior to the measuring session.

The non-linear dynamic effects are remarkable, and even under quasi-static conditions, the response of the system differs significantly when exposed to small ($\pm 25kN$) and large ($\pm 225kN$) load cycles. Unfortunately, under quasi-static conditions, the errors for the large load cycles are substantially more significant than for the small load cycles. While the errors for the large cycles are most important from a fatigue point of view (the large cycles contribute strongly to the fatigue damage), the test speed must be limited to a few load cycles per second. Even at this very low speed an error of the order of 5 % on the large cycles must be accepted.

Again it should be born in mind, that these measurements were taken under the most optimal conditions : a new well maintained system, optimal specimen characteristics and a rather small actuator.

Figure 4 shows the results from measurements on a leaking actuator. The oil leak results in errors even for the static case if the integration term of the servo controller is not activated. The system was loaded by white noise, and the FRF was in this case measured by an FFT analyser. Again we see the non-linear effect resulting in different FRFs for different inputs. The actuator itself is rather fast, and if it was not for the static error caused by the oil leak, the response would be satisfying up to say 30 Hz. The classical way to overcome static error problems is to invoke the integration term on the servo amplifier. The results are shown in figure 4. The response becomes rather poor, and again very low speeds should be used in order to keep the errors at a reasonable level.

Figure 5 illustrates the dependence upon tuning procedure. Measurements were made not on a real system, but on a simple analog simulator (a linear system with one degree of freedom and an eigenfrequency of 67 Hz). Two tuning procedures were used, a time domain procedure and a frequency domain procedure. In the frequency domain procedure the damping (gain of the system) was tuned in order to get a flat FRF, and in the time domain procedure the damping was tuned in order to get a transient-free response to a square wave. The FRF for the two situations are shown in figure 5, and as it is seen, they appear to be quite different. This illustrates the problem of uncertainty in the tuning procedure. Even small changes in the tuning of the system will have a great influence on system behaviour.

2 PRINCIPLES OF CONTROL

It is evident from the above, that there is a strong need for improvement of system response in order to reduce the errors and increase the speeds. In high-cycle stochastic fatigue experiments the numbers of cycles can be very high, and a number of 10^8 cycles is not unrealistic. On a small and fast testing machine it might be possible to run about 10 cycles per second without introducing to large errors. This would, however, give a duration of the test of no less than 3 years.

One way of dealing with this problem is to forget about control and edit the load sequence e.g. as described in the procedures for simulating the IABG-spectrum [4]. Throwing away the smallest cycles reduces the test duration to about $\frac{1}{10}$. If the fatigue problem is a rate sensitive problem e.g. creep fatigue, there are no other ways to reduce the duration of the test than editing the load sequence. In all other cases, however, it is undesirable to do so because of the unknown effects on test results.

The classical way to solve the problem is to try to improve the analog control system and make it fit the circumstances according to the principles of servo control and design of hydraulic systems [5], [6]. To-day, for instance, analog devices with adaptive control are available, but they do not solve the problems with the non-linearities in the system, and it would be nearly impossible to overcome the problem by classical techniques.

An more effective and easy way to reduce the problems of low speed is to use a computer as a kind of intelligent signal generator, as indicated in figure 6.

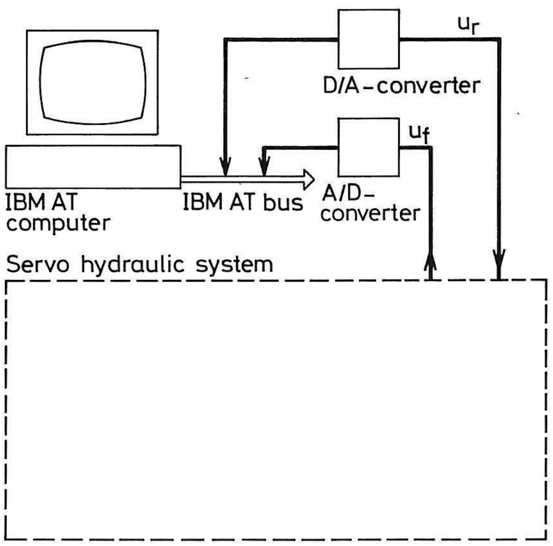

Figure 6. Using a computer as intelligent signal generator.

The idea is to let the computer create a reference signal in such a way that we achieve an acceptable response. First the principles used by Pompetzsky et al. [7] will be outlined. In [7] two principles are mentioned.

3.1 The wait principle and the dynamic switch principle

The first principle, the "wait principle", is very simple and is illustrated in figure 7.a. To apply the next half cycle the computer creates a smooth pulse as reference signal and starts measuring the feedback signal. As soon as the feedback signal is equal to the target level, the computer decides for the next half cycle. The advantages of this concept are that a certain accuracy is assured for all load cycles, and that the system performance is somewhat optimized with respect to accuracy: small cycles run fast, and large cycles run slow. The speed is not increased a lot since the reference signal and the feedback signal are forced to be "in phase". The method is claimed to increase the allowable speed with a factor of about 4.

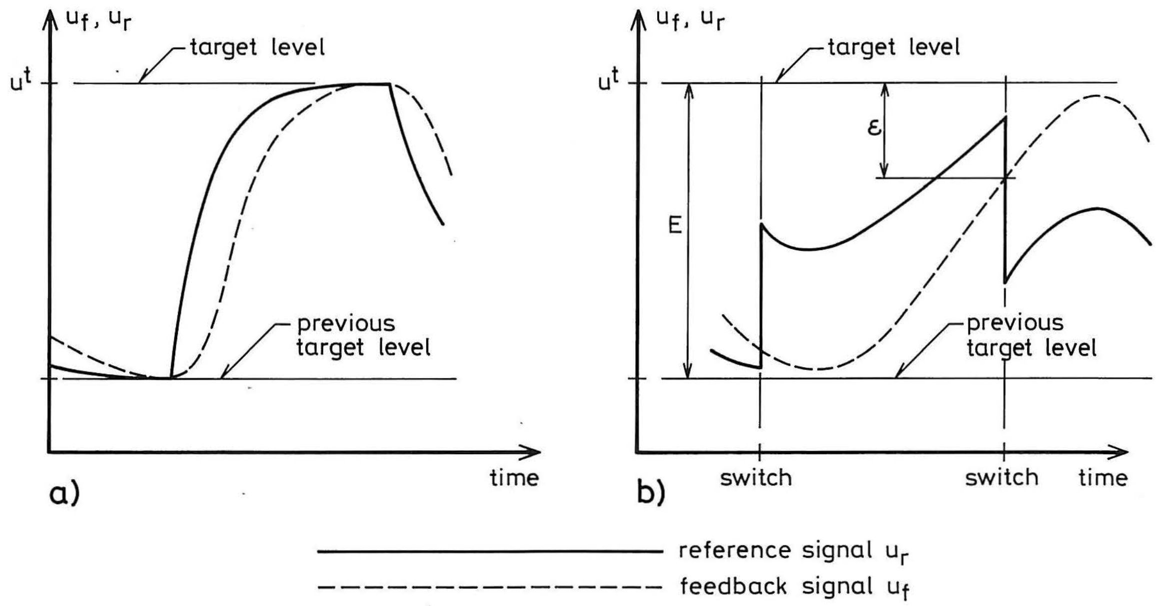

Figure 7. a: the wait principle. b: the dynamic switch principle.

The other principle, the "dynamic switch principle" is allowing the reference signal and the feedback signal to be "out of phase" and is therefore somewhat more effective concerning speed. The dynamic switch principle is illustrated in figure 7.b. To apply a half cycle according to the dynamic switch principle a reference signal is generated in such a way that the error (difference between reference signal and feedback signal) is large. This forces the system to a rapid response because the oil flow, as mentioned above, is proportional to the error. The computer measures the feedback signal, and when the feedback signal is at a certain distance from the target level u^t the next half cycle is applied. The problem here is to know when to switch to the next cycle. The idea in the dynamic switch principle is to switch when the feedback is measured to be greater than or equal to the value u^s obtained from

$$u^s = u^t - \epsilon \tag{4}$$

where u^t is the target level, and ϵ is the so-called threshold value. The threshold value ϵ is assumed to depend only on the excursion E, that is the difference between the actual and the previous target level, and the threshold values are adjusted dynamically in order to make the feedback hit all the target levels as accurately as possible. The target levels are discretized into a limited number of possible values $u_i^t, i = 1, 2...N$. The excursions will then adopt the values $E_j, j = 1, 2...N$ and similarly the threshold values will be a one-dimensional quantity $\epsilon_k, k = 1, 2...N$.

The dynamic switch principle is quite effective concerning speed, and the principle is claimed to increase the speed a factor of about 10. The errors are reduced, and all the peaks and valleys in the feedback history are measured, which makes it possible to store information about the errors. The principle has a few drawbacks however. One is that the principle does not account for static errors exceeding a certain fraction of the threshold values. This means, that if static errors of importance are present in the system, the integration term in the servo amplifier has to be applied resulting in a slow response. However the probably most serious disadvantage is that the principle does not correct for all non-linearities. Since the threshold values are assumed to depend on the excursions only, the absolute values of the actual and the previous target level are not affecting the control procedure, and therefore the non-linearities caused by the absolute values of the target levels will remain in the system. Another disadvantage is that the speed cannot be adjusted; the system runs at its ultimate speed. The speed can be slowed down only by reducing system performance e.g. by reducing the proportional gain. This can only be done within certain limits, since reduction of the proportional gain will increase the static errors in the system. Lastly the principle requires a high computer speed since the decision for switching to the next half cycle is based on simultaneous measurements. This means that the computer measurement and control loop has to run at a frequency that are orders of magnitude higher than the test frequency.

The principle that is the main subject of this paper, the adapting pulse (AP) principle, is more flexible than the principles mentioned above. It goes as fast or even faster than the dynamic switch principle, it is accurate and take account of all non-linearities in the system, the speed can be adjusted without changing the system behaviour, and it requires much less computer speed than the dynamic switch principle.

3.2 The AP principle

The principle is illustrated in figure 8. As for the dynamic switch principle the target levels are discretized into a number of possible values 1,2...N. The computer applies a half cycle by outputting a single output value u^* to make a square pulse with a certain duration T. The duration of the square pulse is determined by

$$T = T_0 + T_1 + t^* \tag{5}$$

where T_0 is the time spend by the computer to make the necessary calculations, T_1 is a user-defined waiting time, and t^* is a waiting time that depends on the prescribed loads. The user-defined waiting time T_1 is the same for all half cycles. The two parameters describing the pulse u^* and t^* , are taken from two N×N matrices

$$\overline{\overline{T}} = \begin{pmatrix} t_{11} & t_{12} & . & . & t_{1N} \\ t_{21} & t_{22} & . & . & t_{2N} \\ . & . & & & . \\ . & . & & & . \\ t_{N1} & t_{N2} & . & . & t_{NN} \end{pmatrix} \tag{6}$$

$$\overline{\overline{U}} = \begin{pmatrix} u_{11} & u_{12} & . & . & u_{1N} \\ u_{21} & u_{22} & . & . & u_{2N} \\ . & . & & & . \\ . & . & & & . \\ u_{N1} & u_{N2} & . & . & u_{NN} \end{pmatrix} \tag{7}$$

where u_{ij} and t_{ij} are the square wave parameters if the actual target level is level i and the previous target level is level j. The matrix elements u_{ij} and t_{ij} are adjusted dynamically in order to minimize the errors at the feedback extremes. As the dynamic switch principle the AP principle is taking advantage of the fact, that large errors increase the speed, and that only the extremes of the feedback signal are of importance. The switch time does not, like for the dynamic switch principle, rely on simultaneous measurements of the feedback signal, but relies on previous experience. The measurements of the feedback signal can therefore be made in a buffer-oriented way, to take advantage of the on board processor of the A/D board and the direct memory access (dma) technique that is available on most data acquisition boards to-day. This means that the measurement and control loop in the computer programme only has to run at the test frequency, and therefore much less computer speed is needed. Furthermore, since a full "from-to" concept is used, all system nonlinearities are eliminated. The principle is able to run both at low and high speeds. The user controls the speed by choosing the user-defined waiting time. If the speed is slow, the user might want to use a smoother shape of the pulse in order to prevent overshoot of the feedback signal. This is easily done by applying a suitable analog low-pass filter between the D/A and the servo hydraulic system.

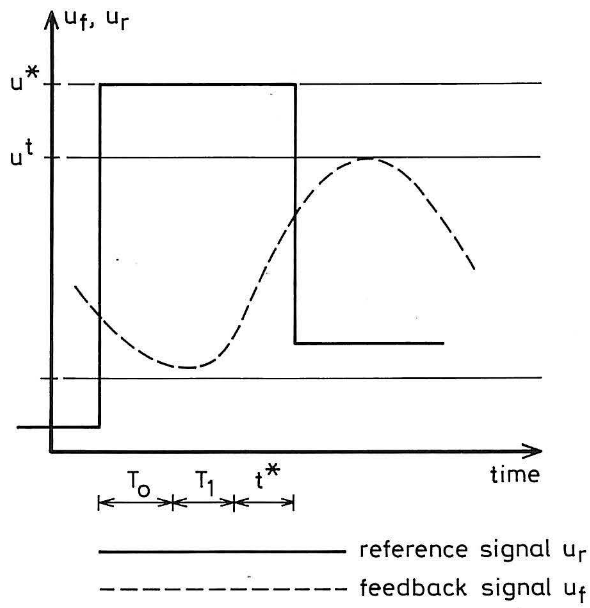

Figure 8. The adapting pulse (AP) principle.

In fact it is not enough to use a "from-to" concept in order to ensure an accurate computer control. An accurate control would in fact require information not only about the actual and the previous target level, but about the whole load history. The reason why it works nicely, as will be shown later, even when only a "from-to" concept is used, is because the system has a rather short memory due to the strong damping. The impulse response function h(t), which describes the memory of the system, is the inverse fourier transform of the frequency response function given by

$$h(t) = \int_{-\infty}^{\infty} H(\nu) e^{i2\pi\nu t} \, d\nu \tag{8}$$

and can be estimated numerically by using the FFT technique, see Brigham [8]. The impulse response functions corresponding to the FRFs for the analog simulator, figure 5, were calculated, and are shown in figure 9. If we define the memory as the time T_m given by

$$\frac{1}{10} max(h(.)) = h(T_m) \tag{9}$$

we find, that for the analog simulator, the memory can be estimated as

$$T_m \in [0.6; 0.8] \frac{1}{\nu_0} \tag{10}$$

where ν_0 is the eigenfrequency of the system. Since the two FRFs for the analog simulator can be considered as tuning bounds, the result given by eq. 10 can be claimed to be a universal estimate for the memory of servo hydraulic loading systems, here interpreting ν_0 as the lowest eigenfrequency of the system. We would expect the control system to work nicely as long as the memory of the system is shorter than the time for performing one load cycle (two half cycles). This means that we would expect the errors to be small as long as the speed (number of cycles pr second) does not exceed approximately $1.25\nu_0$.

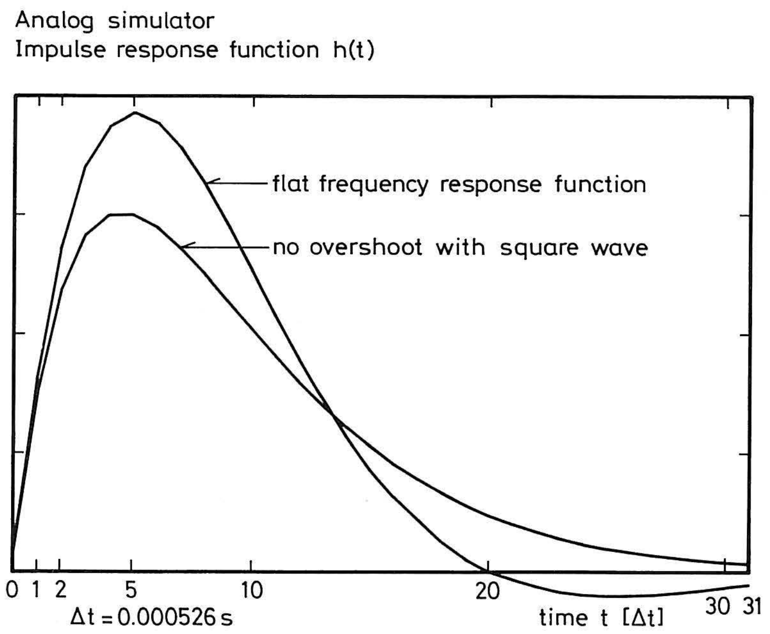

Figure 9. Impulse response functions and their dependency upon tuning procedure.

4 IMPLEMENTATION OF THE AP PRINCIPLE

The principle was programmed in the C programming language and implemented on an IBM AT personal computer with a DT 2821 D/A and D/A board using the Datatranslation library ATLAB for programming the board. The load range was conditioned to $\pm 10V$, and the board has a 12 bit resolution, which maps the $\pm 10V$ range on the integers from 0 to 4095 using offset binary mode. The number of target levels was for simplicity choosen as 8 only. We then have four pictures of the feedback signal, the load picture, the voltage picture, the binary picture and the target picture. The four pictures and the relationships between them are shown in figure 10.

Before starting the control loop, the $\overline{\overline{U}}$ matrix was initialized with the target levels, and the $\overline{\overline{T}}$ matrix was initialized with zeroes.

The extremes of the feedback signal were measured using a buffer technique with dma in order to store a suitable number of measurements in memory without using the IBM AT processor. The core of the program, the control loop, is shown in figure 11. Figure 11.b shows the part of the control loop used for creation of peaks. Creation of valleys take place in a quite similar manner.

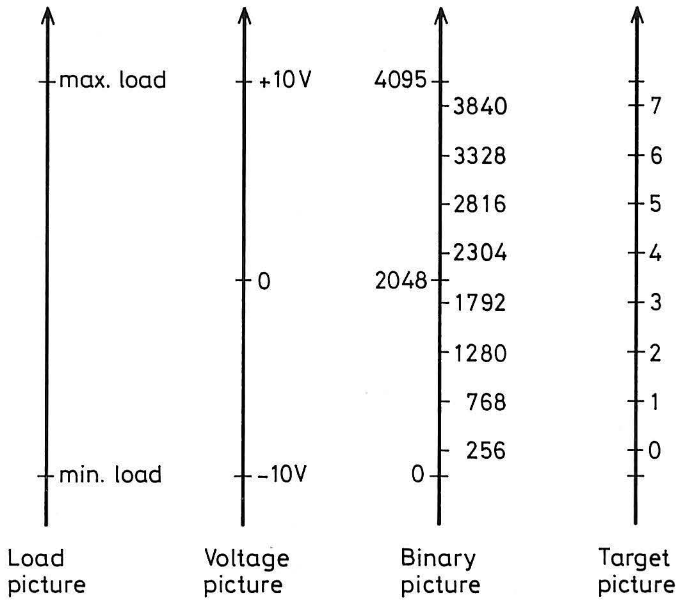

Figure 10. The four feedback pictures.

First the simultaneous simulation of the load process is made according to "the simple" method and the "wide" load spectrum described in [3]. With 8 target levels the simulation is made using two 8×8 matrices $\overline{\overline{K}}^p$ and $\overline{\overline{K}}^v$ containing the jumps given that the next extreme is a peak and valley, respectively. The row entry is the present target level (which is a valley, because peak is to be approached), and the column entry is a random number integer r ranging from 0 to 7.

The random number r is obtained in the following way. First a random integer R is obtained from a mixed congruential pseudo random number generator of the form

$$R_{n+1} = (69069 R_n + 1) \, mod \, 2^{32} \tag{11}$$

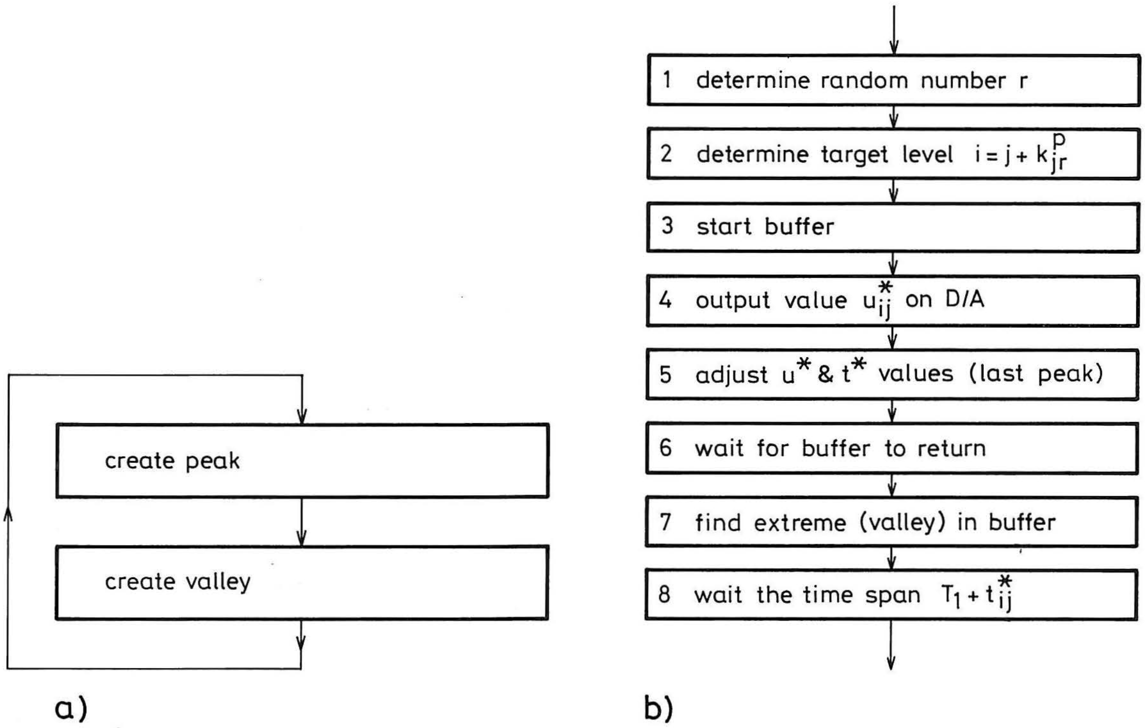

Figure 11. a: the main control loop. b: the valley-to-peak control loop.

where R_{n+1} and R_n are 32 bit integers. The period can be determined as $2^{30} = 1.07 \times 10^9$, see Hammersley [9]. The pseudo random number r was then taken as the three most significant bits of the 32 bit quantity R . This procedure ensures that r has the same long period as R. Once the number r is simulated, the next peak target level i is determined by adding the number j of the present level to the jump k^p_{jr} taken from the valley-to-peak jump matrix as indicated in statement 2 of the loop in figure 11. The valley to peak jump matrix for the "wide" spectrum taken from (3) was found as

$$\overline{\overline{K}}^p = \begin{pmatrix} 1 & 2 & 3 & 4 & 4 & 5 & 6 & 7 \\ 1 & 1 & 2 & 2 & 3 & 3 & 4 & 5 \\ 1 & 1 & 1 & 2 & 2 & 2 & 3 & 4 \\ 1 & 1 & 1 & 1 & 1 & 2 & 2 & 3 \\ 1 & 1 & 1 & 1 & 1 & 1 & 2 & 3 \\ 1 & 1 & 1 & 1 & 1 & 1 & 1 & 2 \\ 1 & 1 & 1 & 1 & 1 & 1 & 1 & 1 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{pmatrix} \qquad (12)$$

As soon as target peak level j is determined, measurements are taken starting the buffer, and the output value u^*_{ij} is converted by the D/A. The buffer just started is storing measurement taken for determination of the present valley. The previous buffer was storing measurements for determination of the previous peak, and the information from these measurements are now made available to future control by adjusting the u^*_{ij} and the t^*_{ij} values in the following way. If the target for the last peak was level l, and the observed peak was u_e, then the u^*_{lj} value is adjusted by a statement like

$$u^*_{lj} = u^*_{lj} + g(u^t_j - u_e) \tag{13}$$

where the constant g is a kind of digital gain. Values for the gain factor g were ranging from 0.2 to 0.5. Adjustments according to eq. 13 are made only if the u^*_{lj} value is not to close to the boundaries, i.e. the u^*_{lj} value must remain within the range [0;4095] after the adjustment. If this is not so, nothing is done to adjust the u^*_{lj} value, and the t^*_{lj} value is increased instead. This means that the durations of the square pulses are used for the coarse tuning and the output values for the fine tuning.

When the adjustments are made, the computer awaits the return of the present buffer, and once it has returned, the extreme is searched for. Now all the necessary procedures for the half cycle have been made, and the computer only has to wait the time span $T_1 + t^*_{ij}$. The total duration of the pulses is determined by eq. 5, and it is seen that the total duration of the pulses will depend on the time used in the control loop. This means that processing times for the control loop becomes a part of system characteristics, and therefore we have to assure that it is well defined. In order to do so all branching statements in the control loop were "balanced", which means that for instance an if-statement was written as

$$\begin{array}{l} if\ condition \\ then\ statement1 \\ else\ statement2 \end{array} \tag{14}$$

where it by the use of dummy statements was ensured that the processing time for statement1 was the same as for statement2.

5 PERFORMANCE AND ACCURACY

Using an IBM AT, the DT2821 board and simultaneous simulation, the ultimate speed was measured to about 130 load cycles (260 half cycles) per second. However, it should be expected to be much higher. The reason is that about 80 % of the time for one half cycle is spent on setting up the board for the dma buffer measurements (statement 3 in the loop in figure 11). This is caused either by a slow on board processor or by inefficiency in the ATLAB software package. If it was not for this unacceptable waiting time, the ultimate speed would have been higher than 500 load cycles per second. Nothing was done to improve the ultimate speed, however, because 130 load cycles per second would be enough for all laboratory test. No standard hydraulic system is able to go faster than this.

Considering the sequence of observed extremes $X_j^k, k = 1,2..$ at the j'th level u_j^t, a picture as indicated in figure 12 is obtained. In the beginning, the systematic errors are dominant because of the dynamics and non- linearities in the system that are not yet corrected for, but as the time goes, the system matrices $\overline{\overline{U}}$ and $\overline{\overline{T}}$ are identified, the systematic errors are getting small, and the stochastic errors become dominant. In fact the systematic errors were measured to be very small; in the order of 0.1 to 0.2 % of full range as one would expect according to eq. 13, and they seem to be independent of system speed. In the following the accuracy is therefore characterised by the stochastic errors defined by

$$\sigma = \sqrt{var[X_j]} \tag{15}$$

Figure 13.a shows the FRF for an analog simulator with an eigenfrequency of 134 Hz using the time domain tuning procedure. Figure 13.b shows for the same simulator the systematic errors $1-|H|$ versus ν (number of load cycles per second) if no computer control is applied, and the random errors σ versus ν if the AP computer control principle is applied. The errors according to this situation can be claimed to be introduced by the control system itself because of the well-defined behaviour of the analog simulator. As it appears from the results, the random errors are small (less than 10 % of full range) for speeds smaller than, say the eigenfrequency of the system, as indicated by eq. 10. We see that the control principle has a tremendous effect on the performance of the system in the sense that the allowable speed can be increased by at least a factor 10. Here it should be borne in mind, that random errors are of much less importance than the systematic errors.

In figure 14 the corresponding results are shown for the Schenck Hydropuls testing machine using medium size load cycles ($\pm$ 100 kN). Again we se a tremendous improvement of system performance, resulting in an increase in allowable speed for this system at a factor of about 30.

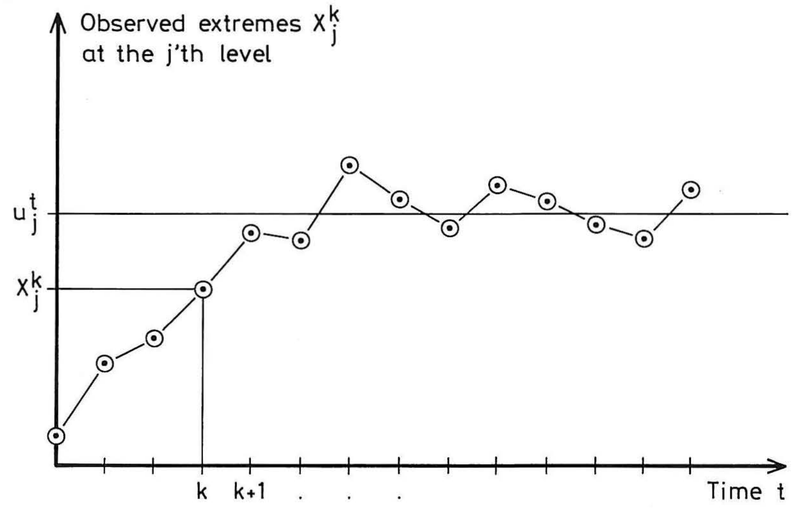

Figure 12. Observed extremes at the j'th level.

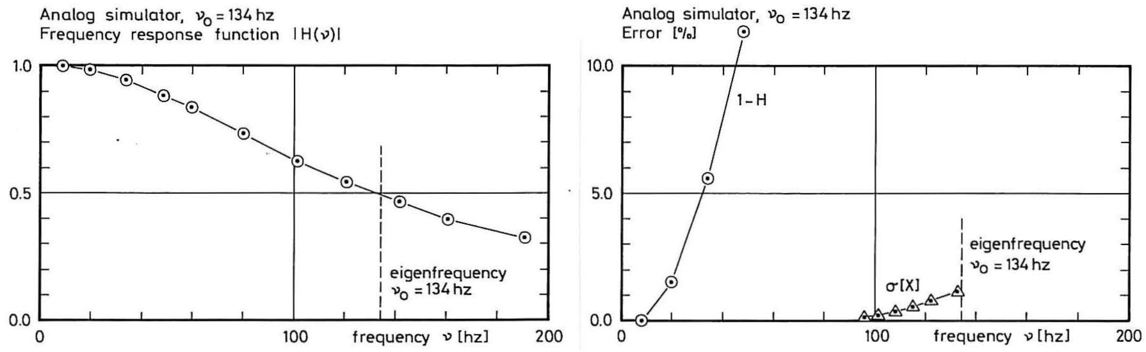

Figure 13. FRF and errors for the analog simulator.

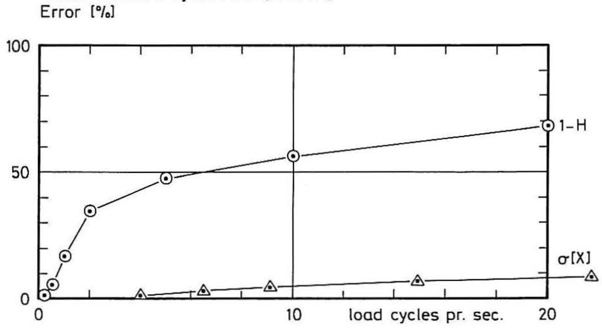

Figure 14. Errors for the Schenck Hydropuls testing machine with and without using the AP principle.

6 CONCLUSION

Typical system errors and lack of performance for traditional servo hydraulic loading systems have been illustrated, and it has been pointed out, that if no measures are taken to reduce system errors, traditional servo hydraulic loading system will give good and reliable test results for stochastic fatigue testing at very low test speeds only.

This is a major experimental problem since the slow speed will give unacceptably long testing times. Traditional servo principles are difficult to apply on usual loading systems because of their strict non-linear behaviour and the crucial demands for speed and accuracy.

A Hybrid concept is proposed where the analog control remains as the basic control mechanism of the system, and a computer acts as a kind of intelligent signal generator in order to reduce remaining system errors. Two different control algorithms proposed in the literature are discussed, and it is concluded, that none of the principles known at present are sufficient to cover all experimental situations satisfactorily.

A new algorithm is proposed that is more effective and requires less computer speed than any of the known algorithms. The use of an IBM AT personal computer with a DT 2821 A/D board gave an ultimate speed of 130 load cycles per second (including simultaneous simulation), but test results indicate that ultimate speeds of about 500 load cycles per second should be possible if a faster A/D board was used. The control principle gave a tremendous improvement of system response since systematic errors were in the order of 0.1 % to 0.2 % of full range, and random errors first became significant at a speed that was 10 to 30 times higher than acceptable speeds for the system if only traditional analog control was used.

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